NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 3001

COMPARISON OF OPERATING CHARACTERISTICS OF FOUR EXPERIMENTAL AND TWO CONVENTIONAL 75-MILLIMETER-BORE CYLINDRICAL-ROLLER BEARINGS AT HIGH SPEEDS

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Lewis Flight Propulsion Laboratory
Cleveland, Ohio

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SUMMARY

An investigation of four experimental and two conventional 75-millimeter-bore (size 215) cylindrical-roller bearings was conducted over a range of DN values (product of bearing bore in mm and shaft speed in rpm) from 0.3×10^6 to 2.3×10^6, radial loads from 7 to 1613 pounds, and oil flows from 2 to 8 pounds per minute with a single-jet circulatory oil feed. The four experimental bearings were equipped with outer-race-riding cages and inner-race-guided rollers. One conventional bearing was equipped with an outer-race-riding cage and outer-race-guided rollers, while the second conventional bearing was equipped with an inner-race-riding cage and inner-race-guided rollers. Each of the six test bearings was equipped with a different design cage made of nodular iron.

The experimental combination of an outer-race-riding cage with a straight-through outer race and inner-race-guided rollers was found to give the best over-all performance based on limiting DN values and bearing temperatures. The better performance of this type bearing over both the conventional inner-race-riding cage type and the conventional outer-race-riding cage type with outer-race-guided rollers is a result of the relative ease of lubrication and cooling and of the adequate oil exiting paths which minimize oil entrapment and churning losses.

The conventional inner-race-riding cage-type bearing could not be successfully operated at DN values above 1.72×10^6 because it is inherently difficult to lubricate and cool. For the same reason, the operating temperatures of this type bearing were higher than those of the four experimental bearings throughout the range of speeds and oil flows investigated.

The conventional outer-race-riding cage-type bearing with outer-race-guided rollers operated successfully at a DN value of 2.1×10^6 but

incurred very severe cage and roller wear at very high speeds, probably because of high cage slip. This type bearing was found to be adequately lubricated and cooled at relatively low oil flows (2 and 2.75 lb/min). At oil flows of 5.5 and 8 pounds per minute, however, this type bearing operated at higher temperatures than the other test bearings because of excessive churning losses.

Cage-pocket type (broached or fitted) had little or no effect on bearing operating temperature or heat dissipation to the oil. Both cages with fitted pockets incurred greater wear in the roller pockets than did their prototypes with broached pockets.

INTRODUCTION

Although cage failures rank high among the causes of bearing failure in high-speed roller and ball bearings in turbojet and turbine-propeller engines (refs. 1 to 6), little research has been reported on either cage materials or cage designs. Basic friction and wear studies of cage materials are reported in references 7 to 9, and an investigation of two cage materials in full-size bearings is reported in reference 10.

As discussed in reference 10, it would be ideal to design a cage with hydrodynamic lubrication at all points of contact between the cage and races and rollers, for then failures due to wear, galling, and cage pickup would not exist. This mode of lubrication has been the object of preliminary NACA research on cage designs, but success has not yet been achieved in this regard. Until hydrodynamic lubrication at all cage-contacting surfaces can be achieved, boundary lubrication will exist at points of sliding contact, and the wear and frictional properties of the cage material will be of extreme importance. In addition, oil interruption requirements are becoming exceedingly severe. For a bearing to operate 15 minutes without oil supply, as is recommended in a recent military specification, auxiliary lubrication systems must be employed to achieve hydrodynamic lubrication. The oil interruption requirement therefore serves to emphasize the cage material problem.

Much work remains to be done on the evaluation of the merits of the more promising cage materials in actual bearings. Data obtained with friction and wear machines merely act as a guide in the choice of materials from which cages should be made, and results cannot be considered conclusive until bearing performance data are at hand.

The evaluation of the relative merits of various cage designs must be considered along with the materials problem. Some of the present cage designs are not satisfactory because they incorporate many inaccessible surfaces making adequate lubrication difficult or impossible. The inner-race-riding cage type, which is perhaps used more widely than

any other, has given fairly good results not withstanding the fact that lubrication of the cage-locating surface is very difficult, since centrifugal force tends to throw the oil away from this surface (ref. 6). There is not much published information on the performance characteristics of outer-race-riding cages. Cages of this type are believed to trap oil, create high churning losses, and thus operate at higher temperatures than do other types of cages. However, such churning losses are due primarily to design and are not necessarily characteristic of this cage type. Furthermore, it appears that a properly designed outer-race-riding cage might prove to be inherently better than other types of cages.

The experimental results reported herein consist of a comparison of the operating characteristics of six 75-millimeter-bore (size 215) cylindrical-roller bearings at DN values (bore in mm times shaft speed in rpm) of 0.3×10^6 to 2.3×10^6, at loads of 7 to 1613 pounds, and at oil flows of 2 to 8 pounds per minute. Test bearings are compared with respect to operating temperatures, limiting DN values, wear, and heat dissipation to the oil. The test bearings, which are described fully in the section TEST BEARINGS, were equipped with nodular iron cages (five outer-race riding and one inner-race riding).

This investigation was conducted at the NACA Lewis laboratory.

APPARATUS

Bearing rig. - The bearing rig (fig. 1) used for this investigation is basically the same as that used in references 11 and 12. Prior to running the tests reported herein, the rig itself was connected to a new gear box and drive motor. In addition, a dashpot was added to the load arm to dampen vibrations of the arm at high speeds.

As shown in figure 1, the bearing under investigation was mounted on one end of the test shaft, which was supported in cantilever fashion in order that bearing component parts and lubricant flow could be observed during operation. A radial load was applied to the test bearing by means of a lever and dead-weight system in such a manner that the loading of the test bearing was essentially unaffected by small shaft deflections or by small shaft and load-arm misalinements.

The support bearings were lubricated in the manner described in reference 11. Oil was supplied to the support bearings at a pressure of 10 pounds per square inch through 0.180-inch-diameter jets and at a temperature equal to that of the test-bearing oil (100° F).

Drive equipment. - The high-speed drive equipment consisted of a shunt-wound 30-horsepower direct-current motor connected to a 14:1 speed

increaser. The high-speed shaft of the speed increaser was connected to the test shaft by means of a floating spline coupling. The speed range of the test shaft was 1100 to 36,000 rpm, controllable to within ±50 rpm at all speeds.

Temperature measurement. - The method of temperature measurement is described in reference 11. Iron-constantan thermocouples were located in the outer-race housing at 60° intervals around the outer-race periphery at the axial midpoint of the bearing under investigation. A copper-constantan thermocouple was pressed against the bore of the inner race at the axial midpoint of the test bearing. The thermocouple electromotive force was transmitted from the rotating shaft by means of small slip rings located on the end of the test shaft (ref. 13).

Lubrication system. - The lubrication system used was of the circulating type. Separate pumps were used to supply oil to the support bearings and the test bearing, and full-flow filters were provided after the oil supply pumps. Oil inlet temperature was controlled to within ±1° F and oil inlet pressure to within ±0.5 pound per square inch. Support-bearing oil was drained by gravity from the base of the rig. Test-bearing oil was collected in cans and pumped either to weighing buckets or back to the sump.

TEST BEARINGS

Six test bearings (size 215) were used for this investigation, the dimensions of which were: bore, 75 millimeters; outside diameter, 130 millimeters; and width, 25 millimeters. Standard SAE 52100 steel races and rollers of the same surface finish and hardness (within manufacturing tolerances) were used. Schematic drawings of all six test bearings are shown in figure 2. All test-bearing cages reported herein and the nodular iron cages used in reference 10 were cut from the same billet. The physical properties and the chemical composition of the cage material are given in table I. Bearings A, B, C, D, and E all had double-flanged inner races (inner-race-guided rollers) and straight-through outer races. Bearing F had a double-flanged outer race (outer-race-guided rollers) and a straight-through inner race. Bearing A was equipped with a one-piece inner-race-riding cage with 20 broached pockets (fig. 2(a)). Bearing B was equipped with a two-piece cage with 16 fitted pockets which was piloted on the outer race outside the roller track (fig. 2(b)). Bearing C was equipped with a one-piece cage with 16 broached pockets which was piloted on the outer race outside the roller track (fig. 2(c)). Bearing D was equipped with a one-piece cage with 16 broached pockets which was piloted on the outer race in the roller track (fig. 2(d)). Bearing E was equipped with a two-piece cage with 16 fitted pockets which was piloted on the outer race in the roller track (fig. 2(e)). Bearing F was equipped with a one-piece cage with 20 broached pockets which was piloted on the shoulders of the outer-race flanges (fig. 2(f)).

The fact that bearings A and F had 20 rollers while bearings B, C, D, and E had only 16 resulted in bearings B, C, D, and E having 33 percent more voids between the rollers at the radial location of the oil nozzle. However, the major resistance to the penetration of oil to the interior surfaces in bearing A was the small clearance between the cage and inner race. The straight-through inner race of bearing F allowed placement of the oil nozzle at a slightly smaller radial distance from the shaft center line to increase the voids between the rollers.

The dimensions of the test bearings, including all critical clearances, are given in table II.

PROCEDURE

Order of test. - Each bearing was first subjected to a number of tests at DN values of 0.3×10^6 to 1.2×10^6, loads of 7 to 1613 pounds, and oil flows of 2 to 8 pounds per minute. Each bearing was then run to the maximum speed at which equilibrium temperatures could be obtained at oil flows of 2 to 8 pounds per minute. At the conclusion of the high-speed tests, the bearings were checked for wear by determining the weight loss of each component part. Each bearing was run through the same series of tests so that the total running time on each bearing was approximately equal.

Lubrication of test bearings. - Lubricant was supplied to the test bearings through a single jet having a 0.050-inch-diameter orifice. The oil was directed normal to the bearing face opposite the load zone. The optimum radial position for the oil jet was determined experimentally for each test bearing and was found to between the cage and inner race for all test bearings.

A highly refined nonpolymer-containing petroleum-base lubricating oil used to lubricate the bearings in several current turbojet engines was used to lubricate the test bearings. The viscometric properties of the test oil are shown in figure 3. Data for figure 3 were obtained from daily samples of test oil. Viscosities were obtained by standard laboratory procedures, and the data plotted in figure 3 represent the mean for all the samples of oil.

Oil was supplied to the test bearings at a temperature of 100° F and at pressures of 30 to 405 pounds per square inch, which corresponded to oil flows of 2 to 8 pounds per minute.

Test-bearing measurements. - Measurements of test-bearing component parts were obtained in a constant-temperature gage room. Standard, precision inspection instruments were used to obtain the dimensions of the test bearings given in table II.

Surface finishes of the bearing component parts (obtained using a profilometer) are given in table III, and hardness measurements of bearing component parts (obtained using a Rockwell superficial hardness tester) are given in table IV.

RESULTS AND DISCUSSION

The results of the experimental investigation are presented in figures 4 to 10 and tables II to IV. Bearing performance is discussed and analyzed with respect to bearing temperature, heat dissipation to the oil, bearing wear, and limiting DN value. The latter is defined as the maximum DN at which the bearing will operate at an equilibrium temperature. In defining limiting DN, it would be ideal to add the condition that severe wear does not occur, but it is very difficult to measure wear during an experimental investigation.

DN Value

Effect of DN on operating temperatures. - Figure 4 shows the effect of DN (values to 1.2×10^6) on the outer-race maximum temperature of all six test bearings at oil flows of 2, 2.75, 5.5, and 8.0 pounds per minute. The outer-race maximum temperature of each of the six test bearings is very nearly a linear function of DN value at all four oil flows over this range of DN values. Bearings B, C, D, and E operated at lower temperatures than did bearings A and F under all conditions of operation, the difference in operating temperatures becoming greater with increasing DN. This indicates that the four experimental bearing types were more effectively lubricated and cooled at all four oil flows than were the two conventional designs. Among the four experimental types, bearings D and E (cage-locating pads between the rollers) operated at slightly lower temperatures than did bearings B and C (cage-locating surfaces outside the roller track). However, the average difference in operating temperatures of bearings B and C at a DN of 1.2×10^6 was only 4° F. The cages of these bearings have a slightly greater tendency to trap oil than do those of bearings D and E.

The curves of figure 4 show that bearing A operates at significantly higher temperatures than the four experimental bearings at the two lowest oil flows, while bearing F operates at the highest temperatures at the two highest oil flows. These results indicate that the interior surfaces of bearing A were not properly lubricated and cooled at the low oil flows and that a high-velocity oil jet is required for sufficient oil to penetrate through the small space between the cage and inner race. The results also indicate that while bearing F was adequately lubricated at the two lowest oil flows considerable oil entrapment between the outer-race flanges occurred at the higher oil flows and resulted in heat generation due to oil churning.

It can be seen then that high operating temperatures result with the conventional inner-race-riding cage-type bearing because of its inherent difficulty of lubrication and cooling and with the conventional outer-race-riding cage-type bearing because of its tendency to trap oil and create high churning losses. Bearings B, C, D, and E operated at lower temperatures than did the conventional bearings, because the lubricant had easy access to the interior surfaces plus a low resistance path to exit from the bearing.

Cage-pocket type had little effect on bearing operating temperature, as is evidenced by the fact that the curves of bearings B and C and those of bearings D and E are coincident at DN values to 1.2×10^6.

Effect of DN on ratio of deflected oil flow to transmitted oil flow. Figure 5 shows the effect of DN on the ratio of deflected oil flow (that oil which is collected on the side of the oil jet) to transmitted oil flow (that oil which is collected on the side opposite the oil nozzle) for the six test bearings. The curves illustrate the relative ease of lubricant flow into and through the bearing types under investigation (transmitted oil flow increases with decreasing flow ratio). For all six test bearings, the flow ratio increased with increasing DN because of the greater tendency to throw oil off the face of the bearing at higher rotative speeds. The flow ratio was highest for bearing A, because it offered the greatest resistance to lubricant flow; and lowest for bearing F, because its straight-through inner race offered little resistance to lubricant flow. Since bearings B, C, D, and E had only 16 rollers while bearing A had 20 rollers, the total of all the voids between the rollers was some 33 percent greater in bearings B, C, D, and E. This could account in part for the greater resistance to lubricant flow, but the major resistance to flow in bearing A was the small space between the cage and inner race.

Effect of DN on power rejected to oil. - Figure 6 shows the effect of DN on power rejected to the oil at oil flows of 2 and 2.75 pounds per minute for all six test bearings. The heat dissipated to the oil is lowest for bearing A, because this type of bearing is inherently difficult to cool. This advantage, although small when compared with the heat dissipated to the oil for bearings B, C, D, and E, is gained only at the expense of the higher operating temperatures previously discussed. The heats dissipated to the oil for bearings D and E are only slightly greater than for bearing A and are somewhat less than those for bearings B and C. The higher heats dissipated to the oil for bearings B and C may be due to the greater tendency of their cages to trap oil and create churning losses higher than those in bearings D and E. The curves for bearing F show that its heat dissipation to the oil does not exceed those of the other bearings until DN values of about 0.85×10^6 (approximately 11,000 rpm) are reached. At higher speeds, however, its heat dissipation to the oil becomes significantly greater than those of the

other bearings. These results illustrate the effects of oil entrapment and the consequent churning losses, which do not become significant until high speeds are reached. The combination of a double-flanged outer race and an outer-race-riding cage, not provided with adequate oil-drainage paths, results in excessive oil churning losses at high rotative speeds.

The fact that the curves for bearings B and C and those for bearings D and E are nearly coincident indicates that cage-pocket type (broached or fitted) has little or no effect on the heat dissipated to the oil.

Load

The effect of load on bearing outer-race maximum and inner-race temperatures for all six test bearings is shown in figure 7. As load was increased from 368 to 1613 pounds, outer-race maximum temperatures increased only slightly, while inner-race temperatures increased at a somewhat greater rate. As the load was increased from 7 and 113 pounds to 368 pounds, outer-race maximum temperatures increased sharply because of the decrease in cage slip. At low loads, roller slip occurs between the rollers and inner race, causing the cage to slip or to rotate at a rotational speed somewhat below its theoretical rotational speed. Percentage cage slip is defined as follows:

$$\text{Percentage cage slip} = \left(\frac{N_{CT} - N_C}{N_{CT}}\right) 100$$

where

N_{CT} theoretical cage rotational speed, rpm

N_C actual cage rotational speed, rpm

At the lowest loads, cage slip varied from 7 to 88.5 percent and was responsible for the decreased rate of heat generation. Inner-race temperatures, however, do not always rise as sharply in going from the low-load to the high-load range, because the rolling and sliding contact which exists between the inner race and rollers when cage slip is present apparently generates nearly as much heat as the pure rolling contact which exists at zero cage slip.

Cage slip has been shown to be a cause of high wear in cylindrical-roller bearings (ref. 10); consequently, operation at light loads should be avoided. The cause of the erratic shape of the inner-race temperature curve for bearing D is unknown; excessive vibration accompanied the sharp rise when the load was increased from 613 to 1113 pounds.

High-Speed Operating Characteristics

Effect of oil flow on limiting DN value. - Each test bearing was run to its limiting DN value from a temperature standpoint only (that speed at which the bearing would not operate at an equilibrium temperature or at which bearing temperature rose rapidly, indicating an incipient failure) at oil flows of 2, 2.75, 5.5, and 8.0 pounds per minute. The results shown in figure 8 indicate that, in general, limiting DN values increase with increasing oil flow. The outer-race-riding cage-type bearings exhibited significantly higher limiting speeds than did bearing A. Bearing A would not run at DN values above 1.72×10^6 (23,000 rpm), whereas bearing C was operated successfully at a DN value of 2.32×10^6 (31,000 rpm). The type of cage used in bearing C seems to be best suited to ultra-high-speed operation. Of the four experimental bearings, bearing E showed the poorest high-speed operating characteristics; this may have been due to a weakness in design or to the relatively small diametral clearance in this bearing.

Although bearing F (data from ref. 10) was successfully operated at a DN value of 2.1×10^6 (28,000 rpm), operating temperatures above a DN of 1.5×10^6 were extremely erratic because of high cage slip, which produced extremely high wear as discussed in the section Wear Data.

Effect of DN on operating temperatures at very high speeds. - Curves of bearing outer-race maximum temperature as a function of DN at very high speeds for the six test bearings are shown in figure 9 at oil flows of 2.75 and 8 pounds per minute. It is evident that the linear relation which existed at low speeds (fig. 4) does not hold for all test bearings at very high speeds. The curves for bearings B, C, D, and E are generally linear, but those of bearings A and F are quite erratic and indicative of unstable operation at very high speeds. At an oil flow of 2.75 pounds per minute, the slope of the curve for bearing F showed a marked tendency to increase with speed, probably because of the increasing rate of heat generation due to oil churning. Thus, the basic weakness of this design becomes more apparent at very high speeds. At an oil flow of 2.75 pounds per minute and a DN of 1.65×10^6, bearing F operated at a temperature 54° F higher than did bearing C. The curve for bearing F (data from ref. 10) at an oil flow of 8 pounds per minute is quite interesting in that the bearing temperature decreased when the speed was increased from a DN of 1.5×10^6 to 1.65×10^6. This unusual phenomenon, discussed fully in reference 10, was caused by a sudden increase in cage slip from 1 or 2 percent to 35 percent.

Wear Data

Wear data for the test-bearing component parts (wear indicated by weight loss) are shown in bar graphs on figure 10. These data, together with thorough visual inspections, are invaluable because they reveal the

critical or weak points in the various bearing designs. However, because wear is such a complex phenomenon, the data should be used only qualitatively and not quantitatively.

Heaviest wear in bearing A occurred between the roller ends and the inner-race roller track flanges. In bearing A very little oil gets to this location, because it is difficult for the oil to penetrate through the small space between the inner-race flange and the cage inside periphery; whatever oil does pass through this barrier is immediately thrown outward by centrifugal force.

Heavy wear occurred in bearing B in the cage-roller pockets, with the rollers sustaining especially heavy wear. In contrast, no severe wear occurred at any point in bearing C. Since the cages of bearings B and C were alike except for cage-pocket design, the more intimate contact produced by the fitted pockets of bearing B together with their relatively greater inaccessibility to the lubricant may have contributed to the higher wear. Neither bearing B nor C sustained any significant wear at its cage-locating surface.

In bearing D heavy wear occurred at the cage outside periphery and at the outer-race inside periphery. Apparently, the cage-locating pads of bearing D were not of sufficient size to support loads between the cage and its locating surface, because very little wear occurred at this point in bearing E which had cage-locating pads that were about 60 percent larger than those of bearing D. Bearing E sustained greater wear in the cage pockets than did bearing D. Here again, broached roller pockets seem to be superior to fitted roller pockets.

In bearing F (data from ref. 10) severe wear occurred in the cage-roller pockets. Both cage and roller wear were extreme; these high values of wear were found to be the result of the high cage slip which occurred at DN values above 1.5×10^6 at the higher oil flows (ref. 10). The possible causes of this cage slip are discussed in reference 10.

SUMMARY OF RESULTS

The following results were obtained in an investigation of four experimental and two conventional 75-millimeter-bore (size 215) cage-type cylindrical-roller bearings (each equipped with a different design nodular iron cage) which were operated over ranges of DN values (product of bearing bore in mm times shaft speed in rpm) from 0.3×10^6 to 2.3×10^6, radial loads from 7 to 1613 pounds, and oil flows from 2 to 8 pounds per minute introduced through a single oil nozzle:

1. The experimental combination of an outer-race-riding cage with a straight-through outer race and inner-race-guided rollers was found to

give the best over-all performance based on limiting DN values and bearing temperatures. The better performance of this type bearing over both the conventional inner-race-riding cage type and the conventional outer-race-riding cage type with outer-race-guided rollers was a result of relative ease of lubrication and cooling and of adequate oil exiting paths which minimize oil entrapment and churning losses.

2. Of the two basic types of experimental outer-race-riding cages investigated, those with piloting surfaces outside the roller track were operated successfully at higher DN values (2.32×10^6) than were those with piloting surfaces between the rollers. On the other hand, the latter type of cage operated at slightly lower temperatures, presumably because of reduced oil entrapment and churning losses.

3. The conventional inner-race-riding cage-type bearing could not be successfully operated at DN values above 1.72×10^6 because it is inherently difficult to lubricate and cool. For the same reason, the operating temperatures of this type bearing were higher than those of the four experimental bearing types throughout the range of speeds and oil flows investigated. The heat dissipation to the oil was, however, slightly lower for this type of bearing than for the experimental types.

4. The conventional outer-race-riding cage-type bearing with outer-race-guided rollers operated successfully at a DN value of 2.1×10^6 but incurred very severe cage and roller wear at very high speeds because of high cage slip. This type bearing was found to be adequately lubricated and cooled at relatively low oil flows (2 and 2.75 pounds per minute). At oil flows of 5.5 and 8 pounds per minute, however, this type bearing operated at higher temperatures than the other test bearings because of excessive churning losses. At DN values above 0.85×10^6, the heat dissipated to the oil for this bearing exceeded that of the other test bearings, presumably because of high churning losses. Temperatures and heat dissipation to the oil increased at a greater rate with increasing DN for this type bearing than for the other types investigated.

5. Cage-pocket type (broached or fitted) had little or no effect on bearing operating temperature or heat dissipated to the oil. Both cages with fitted pockets incurred greater wear in the roller pockets than did their prototypes with broached pockets, presumably because of the more intimate roller contact produced by fitted pockets and the greater inaccessibility of fitted pockets to lubricant flow.

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TABLE I. - PROPERTIES AND COMPOSITION OF

NODULAR IRON AS CAST[a]

Yield point[b], psi	56,800
Tensile strength, psi	80,500
Elongation, percent	2
Reduction of area, percent	1.5
Modulus of elasticity, psi	20.9×10^6
Chemical composition, percent	
C	3.70
Mn	.33
Si	2.33
Cr	.02
Cu	.47
P	.03
S	.011
Mg	.061

[a]Data obtained from Ford Motor Co. foundry.

[b]Specimen taken from $2\frac{1}{4}$-in. radius of a 6 in.-diameter by 12-in.-long ingot.

TABLE II. - DIMENSIONS OF TEST BEARINGS

Bearing	A		B		C		D		E		F	
Pitch diameter of bearing, in.	4.036		4.036		4.036		4.036		4.036		4.036	
Length-diameter ratio of rollers	1		1		1		1		1		1	
Number of rollers	20		16		16		16		16		20	
Cage type	One piece, broached pockets, located on inner-race flanges.		Two piece, fitted pockets, located on outer-race inside diameter outside the roller track.		One piece, broached pockets, located on outer-race inside diameter outside the roller track.		One piece, broached pockets, located on outer-race inside diameter between the rollers.		Two piece, fitted pockets, located on outer-race inside diameter between the rollers.		One piece, broached pockets, located on outer-race flanges.	
	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After
Total running time, hr	0	25.9	0	25.1	0	21.9	0	23.7	0	29.7	0	23.2
Average roller diameter, in.	0.5511	0.5505	0.5511	0.5506	0.5511	0.5509	0.5510	0.5509	0.5512	0.5510	0.5511	0.5501
Average roller length, in.	0.5507	0.5505	0.5508	0.5506	0.5507	0.5505	0.5506	0.5505	0.5508	0.5506	0.5507	0.5500
Total roller diametral variation, in.	4.8×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------	5×10^{-5}	------
Total roller length variation, in.	0.0004	------	0.00015	------	0.00045	------	0.0002	------	0.0003	------	0.0004	------
Axial clearance between roller and race flanges, in.	0.0025	0.0055	0.0021	0.0043	0.0022	0.0037	0.0026	0.0047	0.0028	0.0044	0.0022	0.0038
Axial clearance between roller and cage, in.	0.012	0.013	0.0065	0.007	0.006	0.0065	0.007	0.007	0.0065	0.0065	0.012	0.012
Circumferential clearance between roller and cage, in.	0.011	0.015	0.011	0.0145	0.006	0.010	0.009	0.0105	0.013	0.014	0.0115	0.051
Mounted bearing: Diametral clearance, in. Bearing Cage	0.001 .014	0.0012 .0145	0.0009 .008	0.0014 .0085	0.0011 .009	0.0017 .009	0.0008 .008	0.0013 .014	0.0005 .0075	0.001 .008	0.0010 .0205	0.0030 .0225
Eccentricity, in.	0.0001	0.0002	0.0001	0.0002	0.0001	0.0002	0.0000	0.0008	0.0001	0.0002	0.0001	0.0003
Remarks	Heavy wear on roller ends and inner-race flange faces.		Heavy wear in cage-roller pockets and on rollers. Good high-speed operation.		Wear of all parts light. Best performance of all bearings at very high speeds.		Heavy wear on outer-race inside diameter and cage outside diameter.		Heavy wear in cage-roller pockets. Vibration at high speeds.		Heavy wear in cage-roller pockets and on rollers because of cage slip at very high speeds.	

TABLE III. - TEST-BEARING SURFACE-FINISH MEASUREMENTS[a]

Bearing	A		B		C		D		E		F	
	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After
Inner-race roller riding diameter, circumferential	1.5-3	2-4	4	5-14	4	1.5-3	2-4	2-3	2-4	2-3	2-3.5	2.5-20
Roller diameter, axial	2-4	2-6	2-4	2-4	3	1.5-3	1.5-3.5	1.5-3.5	3-4	1.5-3.5	3-4.5	2-6
Roller end	2-3.5	2-4	3-4	3-8	3-4	3-7	3-4	2-15	3-4	2-16	3-4	1.5-10
Cage-locating surface, circumferential	3-5	3-5	3-5	8-12	4-6	2-4	3-4	2-5	3-5	3-6	3-5	2-5
Outer-race roller riding diameter, circumferential	3-4	2-3.5	3-5	4-8	4-6	1.5-2.5	3-4	2-5	3-5	3-6	3-5	1.5-4
Cage, locating diameter	14-26	10-18	15-20	10-20	15-20	10-20	15-20	3-6	15-20	3-6	10-16	15-20

[a] Obtained with profilometer.

TABLE IV. - TEST-BEARING-HARDNESS MEASUREMENTS[a]

Bearing	A		B		C		D		E		F	
	Before	After	Before	After	Before	After	Before	After	Before	After	Before	After
Outer-race face (Rockwell C-scale)	59.5-61	59.5-60.5	60	60-61.5	60	60.5-62	60-61	59-60.5	60-61	60-60.5	61-62	59-61
Inner-race face (Rockwell C-scale)	59-60	60-61	60	60-61.5	61	61-61.5	62	60-61	61-62	60-60.5	62	60-61
Roller end (Rockwell C-scale)	65-65.5	63-65	65	63-63.5	65	63-64	64-66	64.5-66	65	64-65.5	63-65	61-63
Cage face (Rockwell B-scale)	96	93-98	93	92-96	93	95-98	93	93	93	91-93	97	92-95

[a] Obtained with Rockwell superficial hardness tester.

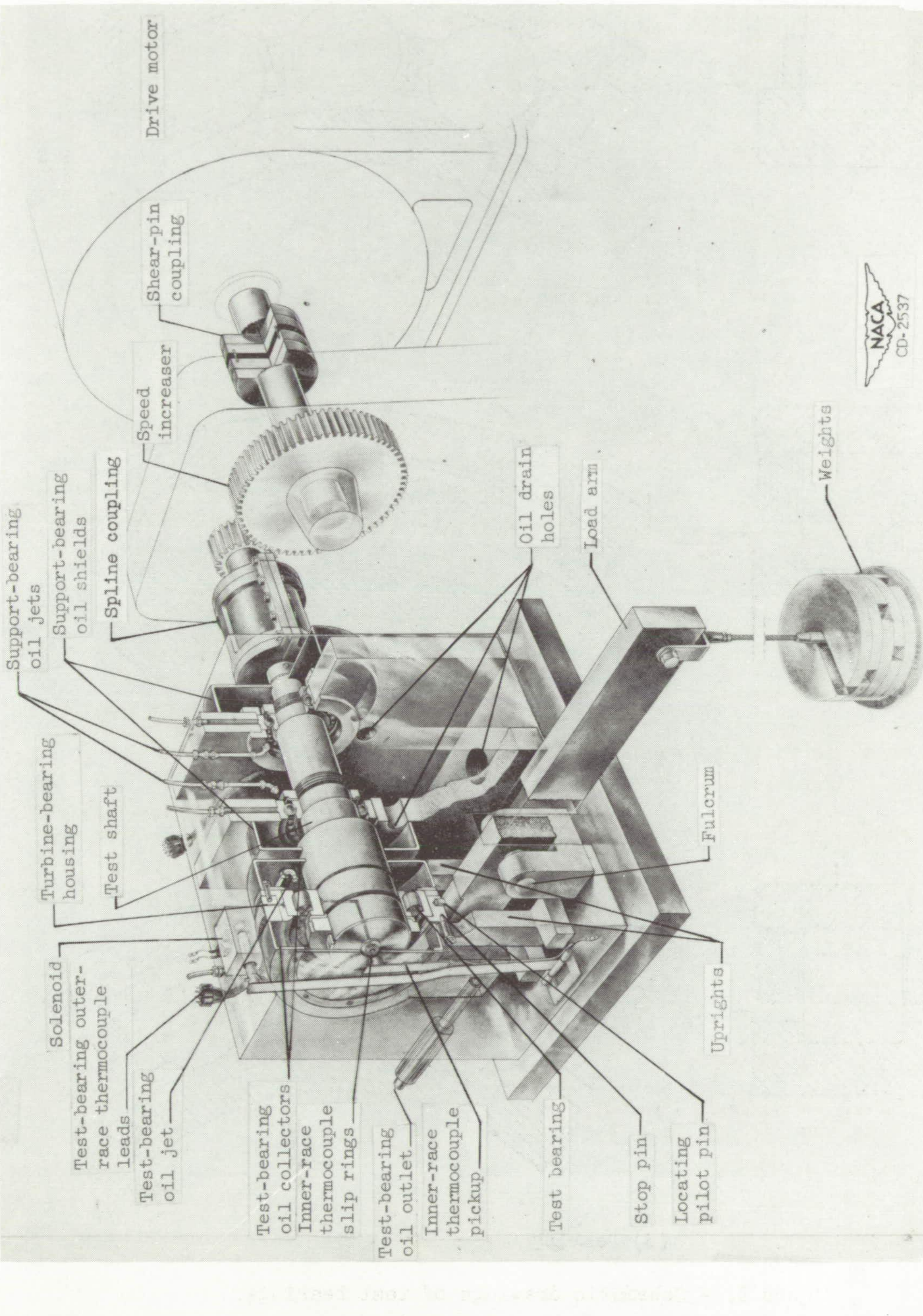
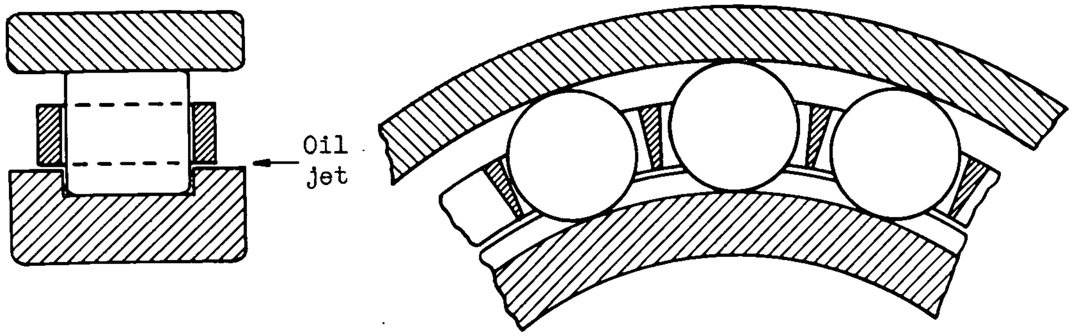
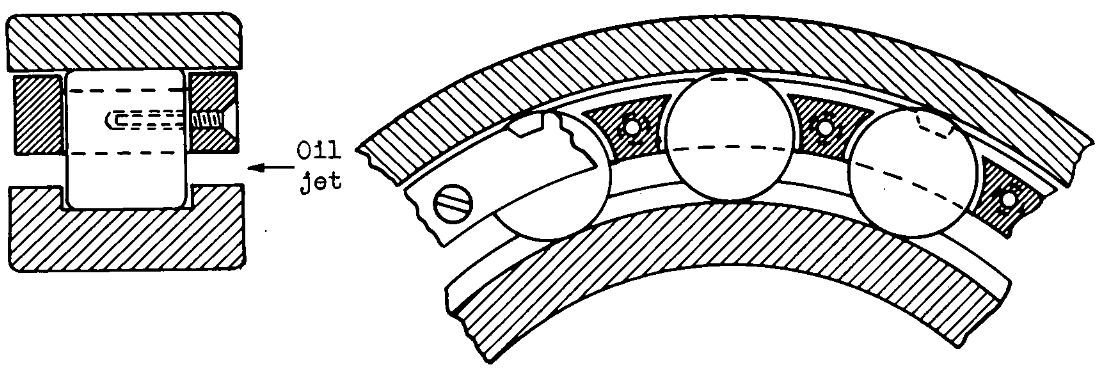


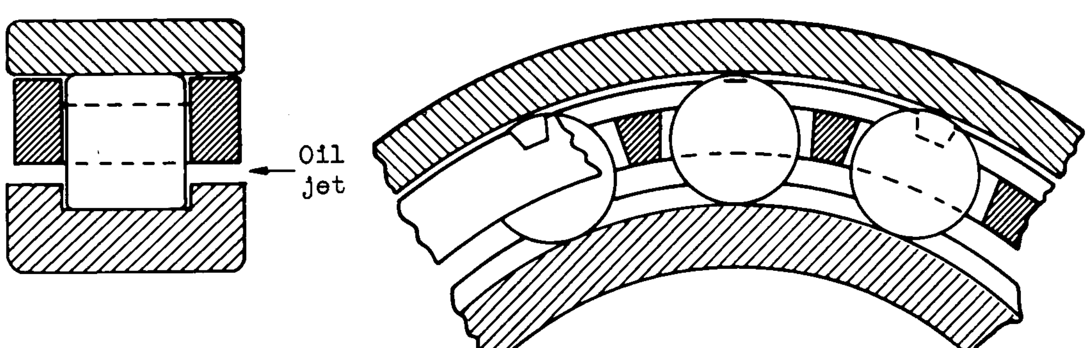
Figure 1. - Cutaway view of radial load rig.



(a) Bearing A



(b) Bearing B



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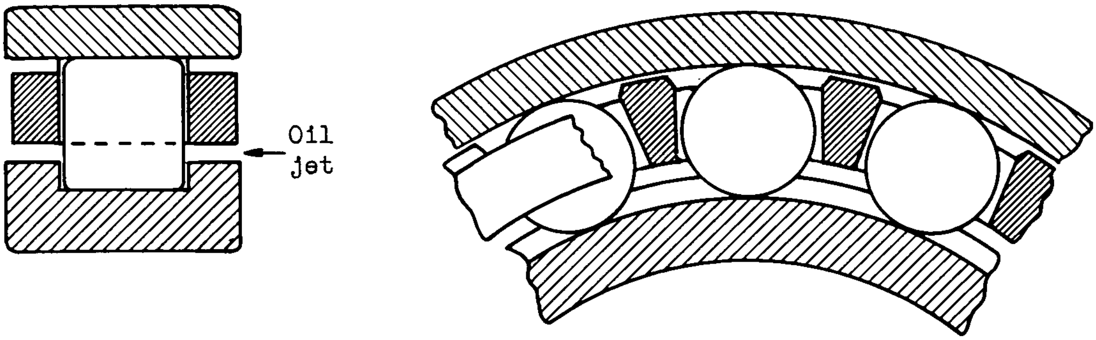
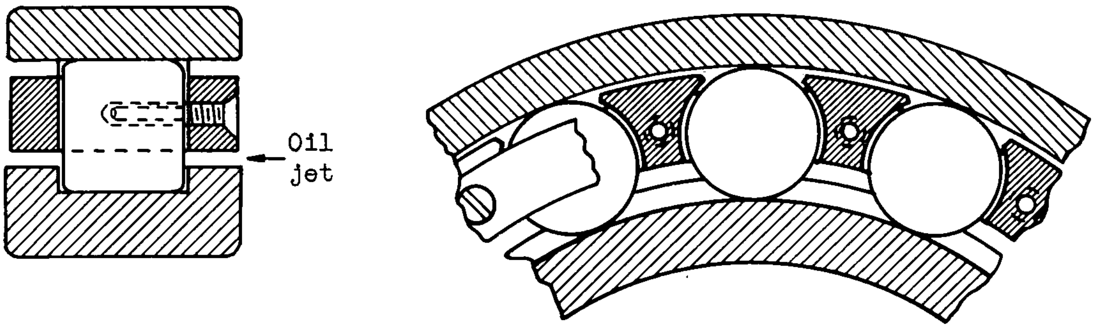
(c) Bearing C

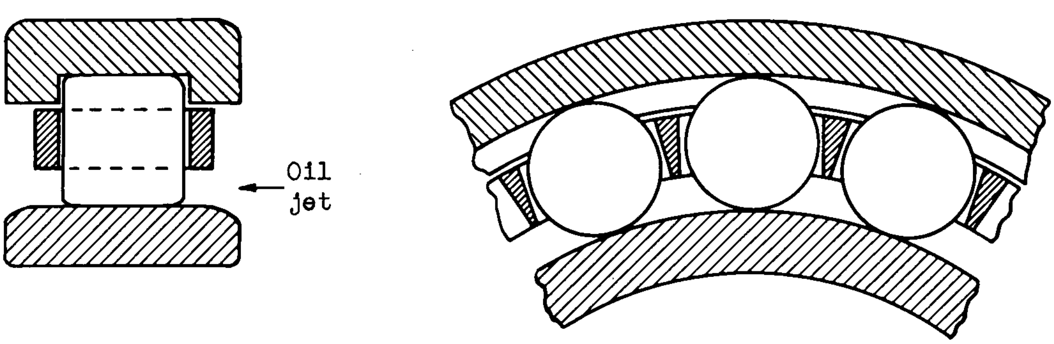
Figure 2. - Schematic drawings of test bearings.



(d) Bearing D



(e) Bearing E



(f) Bearing F

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Figure 2. - Concluded. Schematic drawings of test bearings.

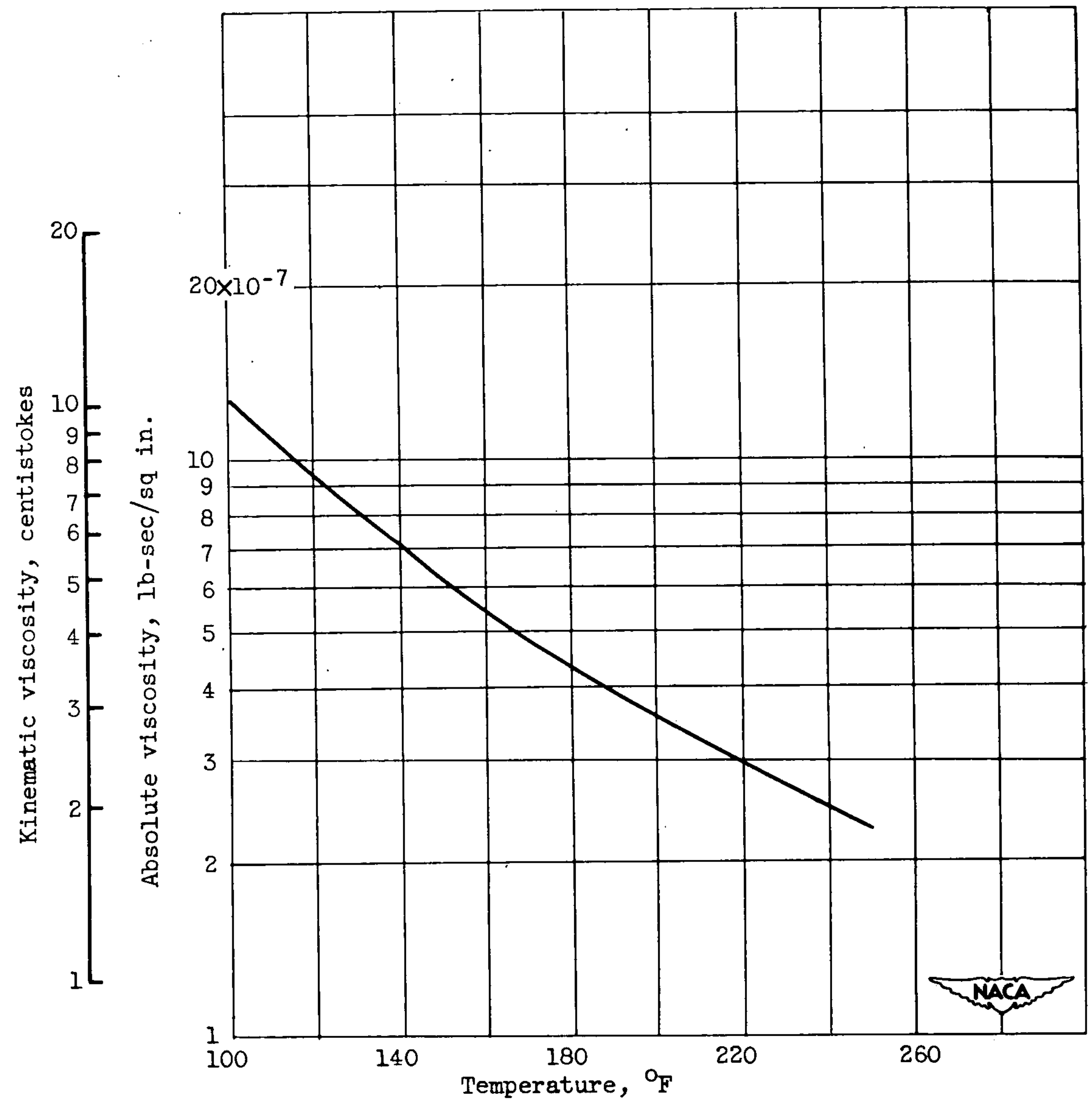
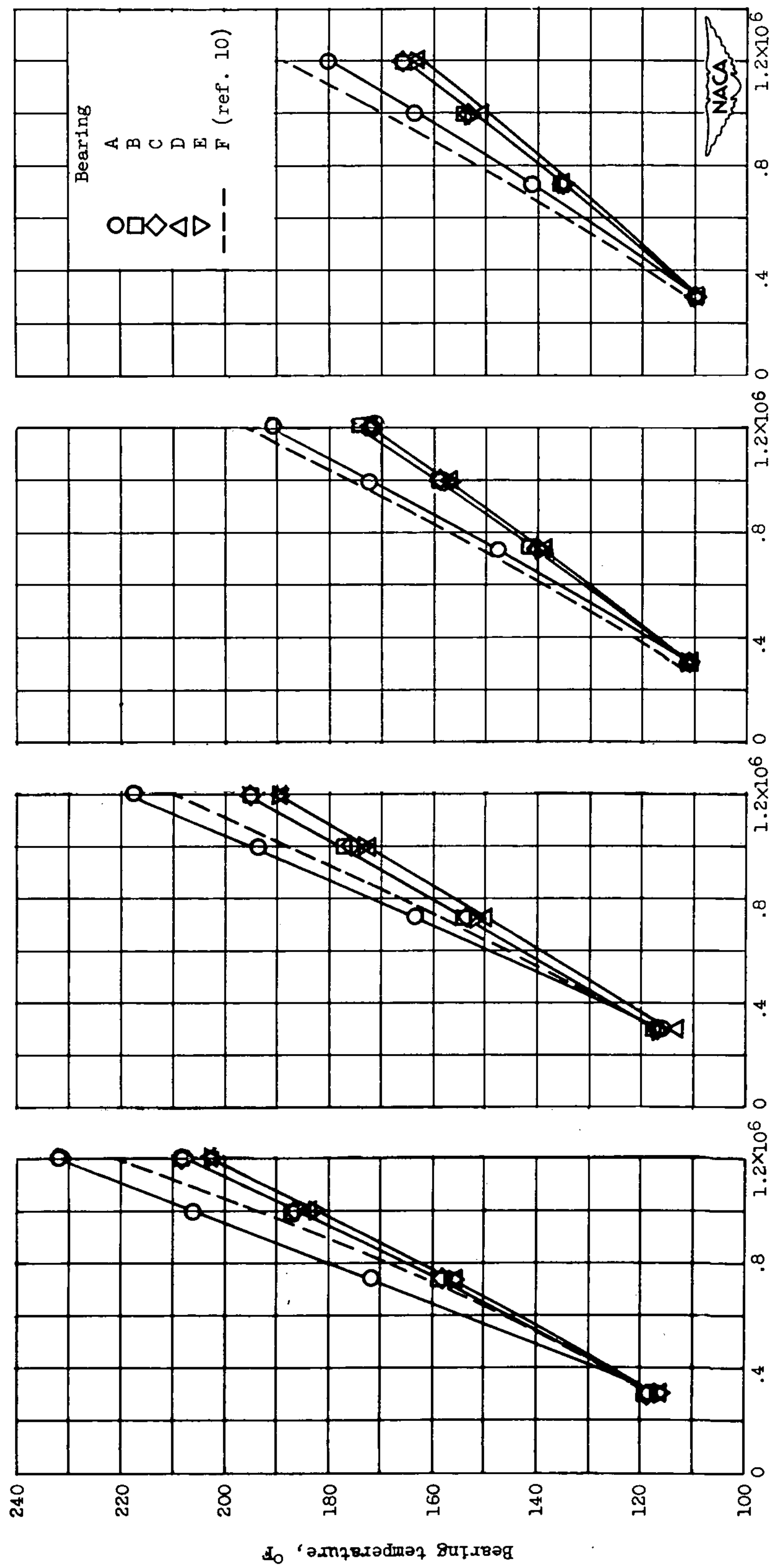


Figure 3. - Effect of temperature on kinematic and absolute viscosities of test oil. Pour point, less than -75° F; flash point, 300° F; viscosity index, 75; autogenous ignition temperature, [a]500° F.

[a]Time lag before ignition at temperature indicated was under 2 min.



(a) Oil flow, 2 pounds per minute.

(b) Oil flow, 2.75 pounds per minute.

(c) Oil flow, 5.5 pounds per minute.

(d) Oil flow, 8.0 pounds per minute.

Figure 4. - Effect of DN on bearing outer-race maximum temperature at four oil flows for bearings A, B, C, D, E, and F. Load, 368 pounds; oil inlet temperature, 100° F.

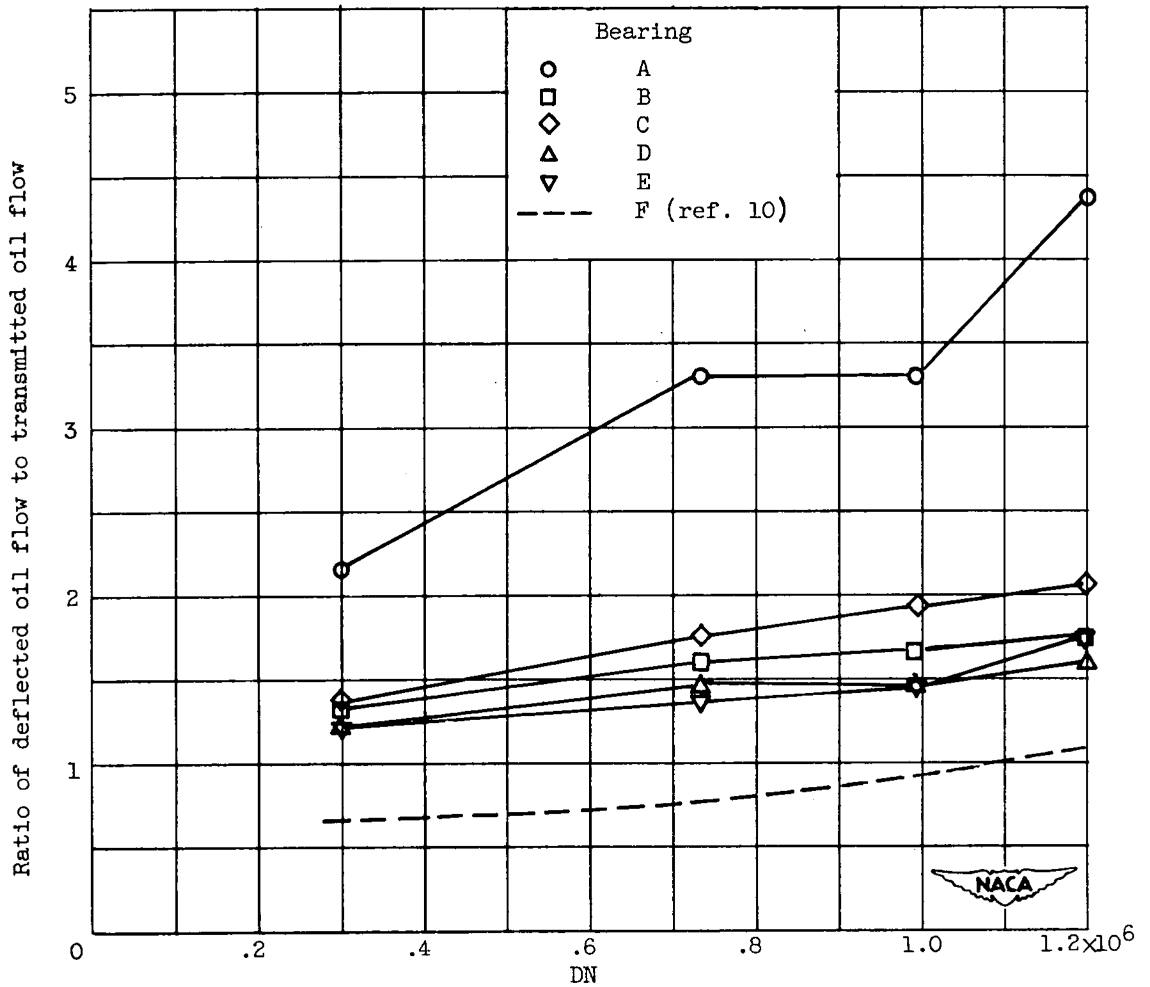
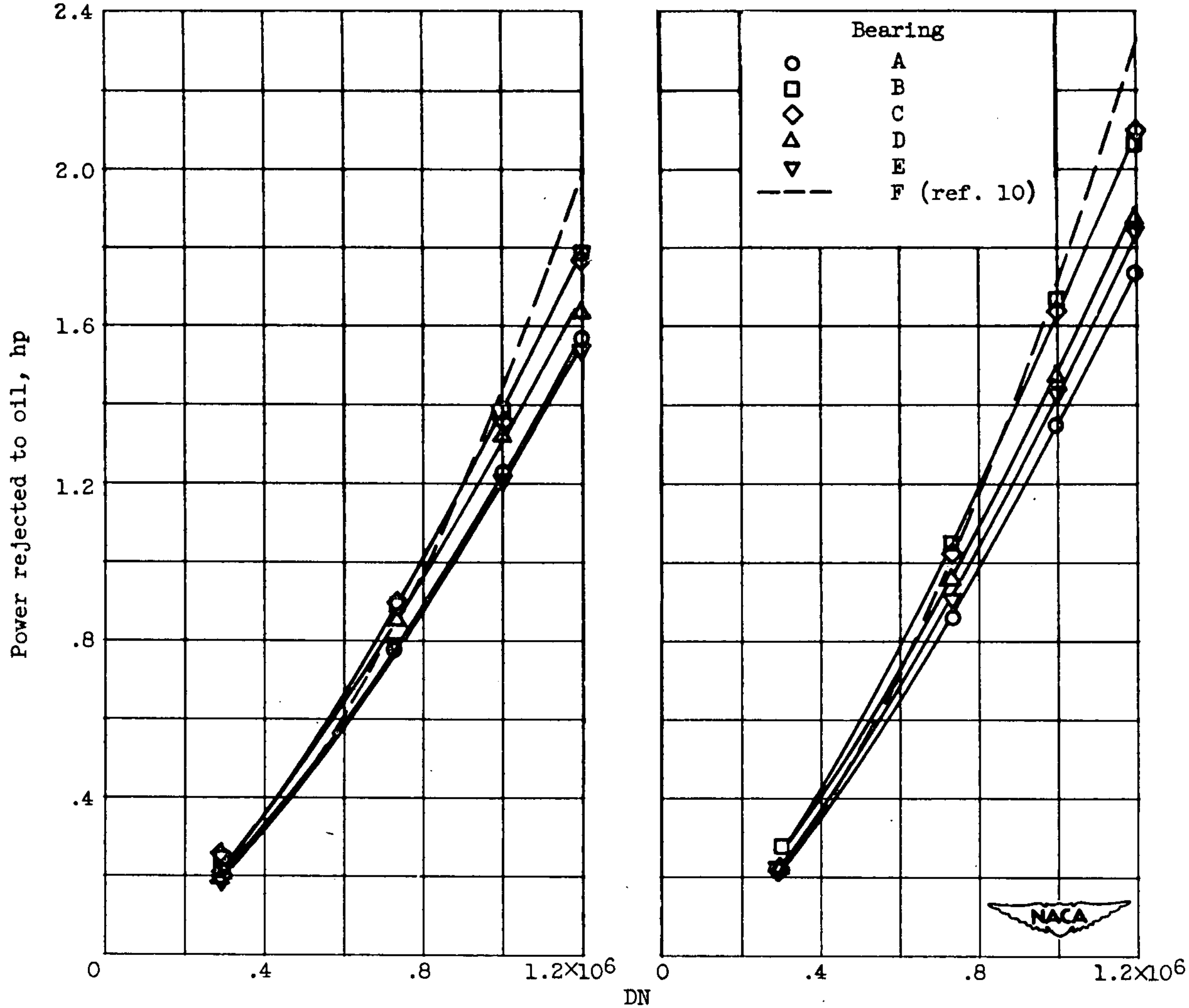


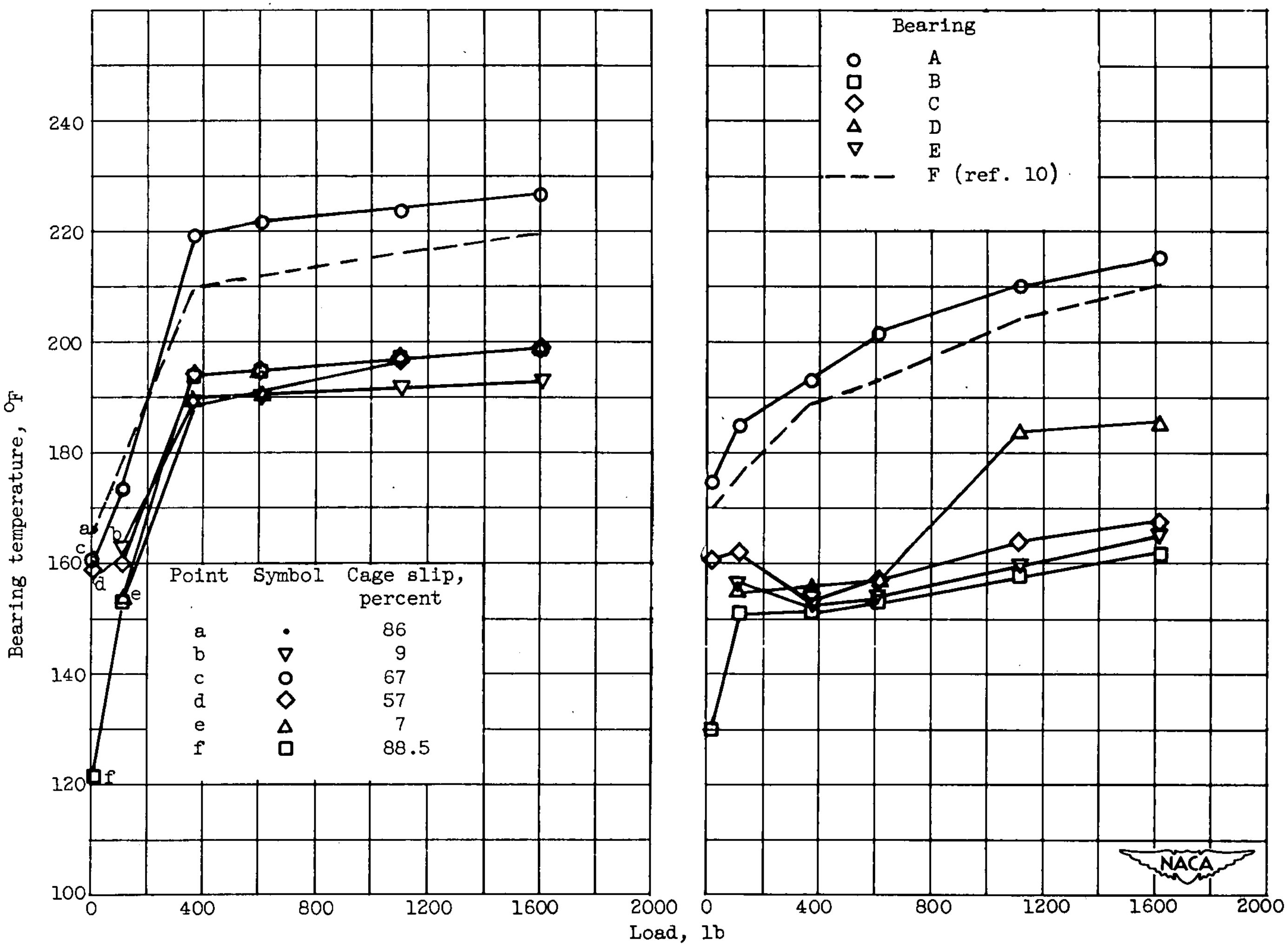
Figure 5. - Effect of DN on ratio of deflected to transmitted oil flows. Load, 368 pounds; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F.



(a) Oil flow, 2 pounds per minute.

(b) Oil flow, 2.75 pounds per minute.

Figure 6. - Effect of DN on power rejected to the oil at two oil flows for bearings A, B, C, D, E, and F. Load, 368 pounds; oil inlet temperature, 100° F.



(a) Outer-race maximum temperatures.

(b) Inner-race temperatures.

Figure 7. - Effect of load on bearing outer-race maximum and inner-race temperatures for bearings A, B, C, D, E, and F. DN, 1.2×10^6; oil flow, 2.75 pounds per minute; oil inlet temperature, 100° F.

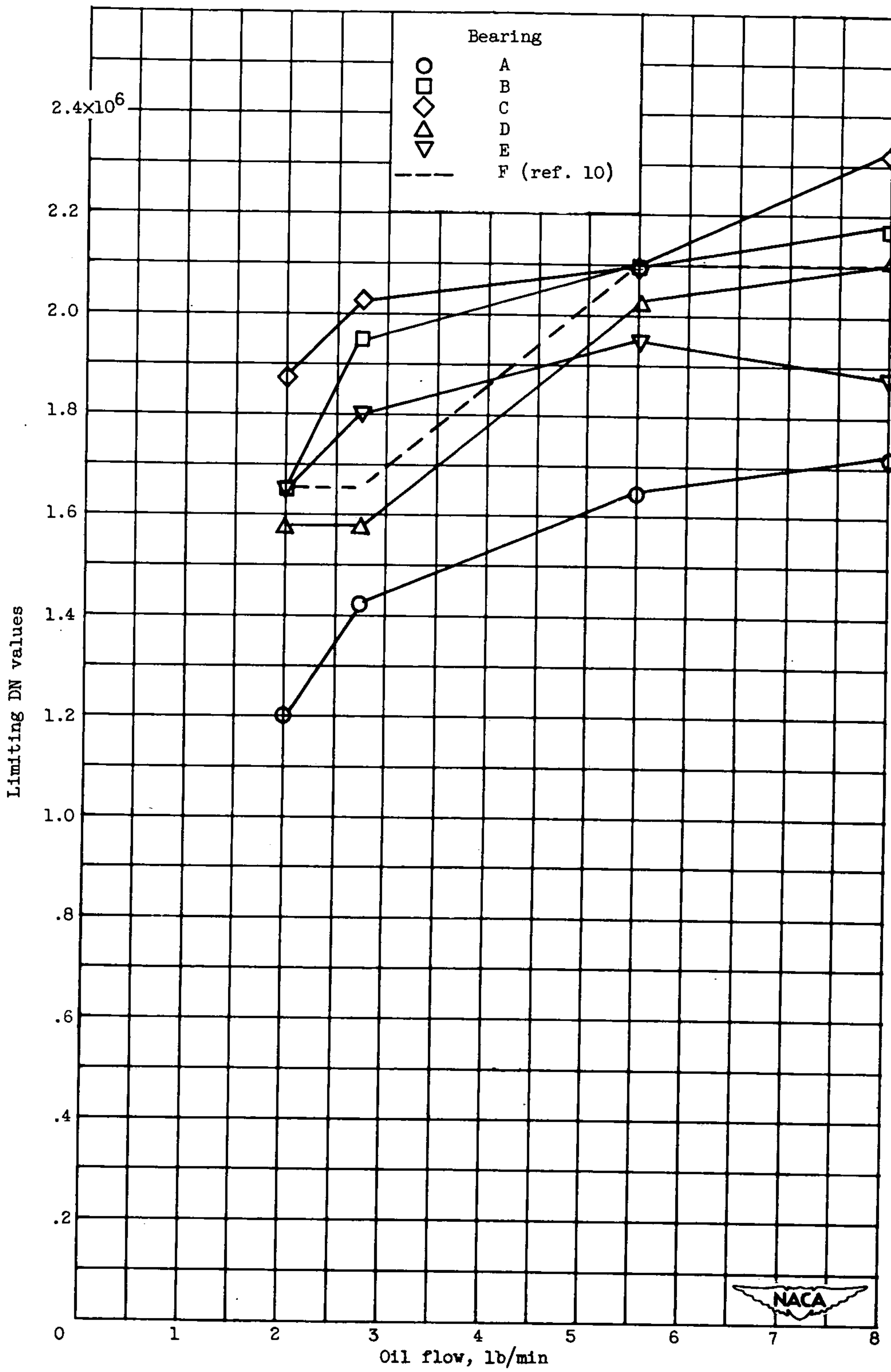
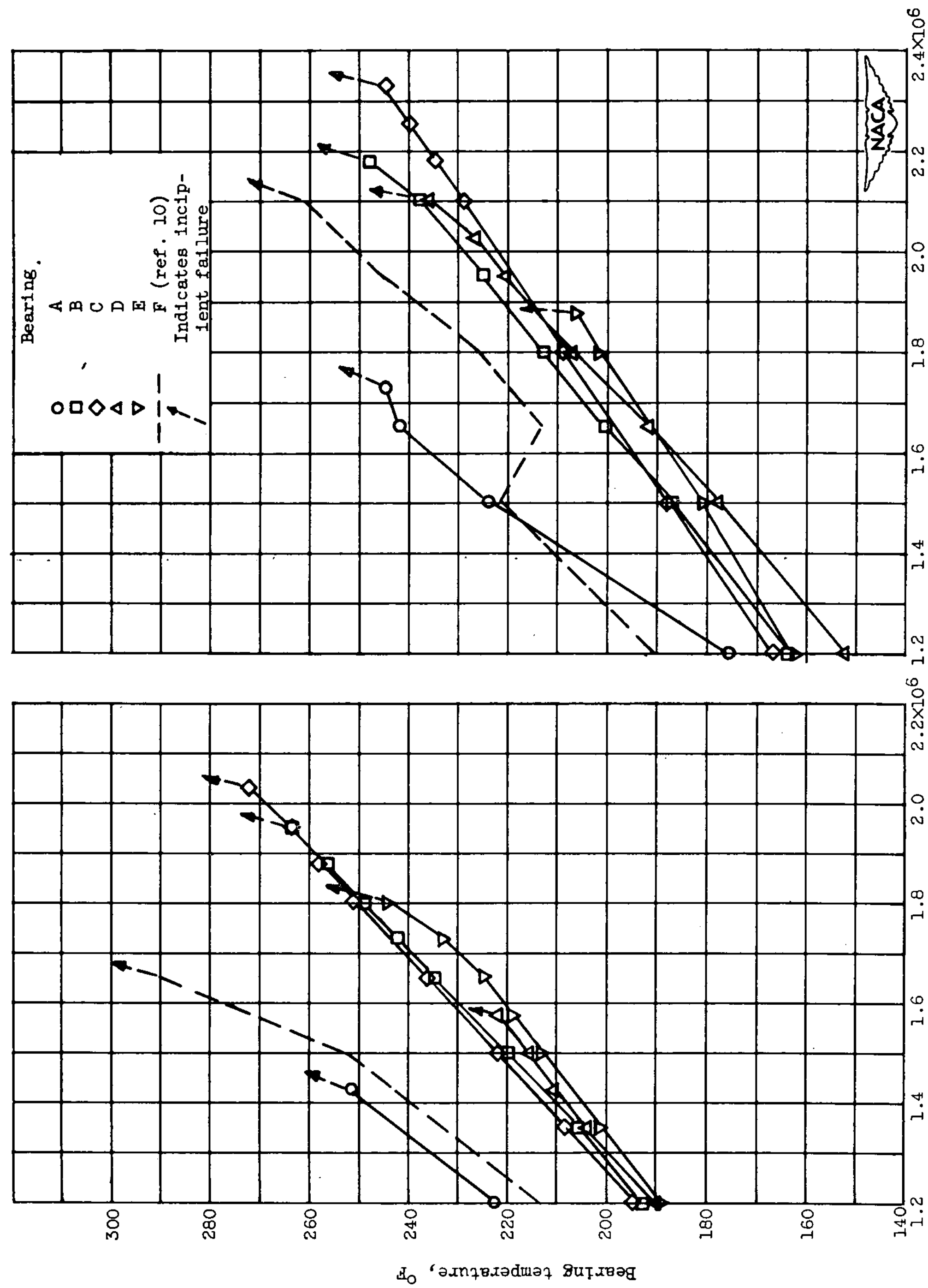


Figure 8. - Effect of oil flow on limiting DN value for bearings A, B, C, D, E, and F. Load, 368 pounds; oil inlet temperature, 100° F.



(a) Oil flow, 2.75 pounds per minute.

(b) Oil flow, 8.0 pounds per minute.

Figure 9. - Bearing outer-race maximum temperature as function of DN at very high speeds at two oil flows for bearings A, B, C, D, E, and F. Load, 368 pounds; oil inlet temperature, 100° F.

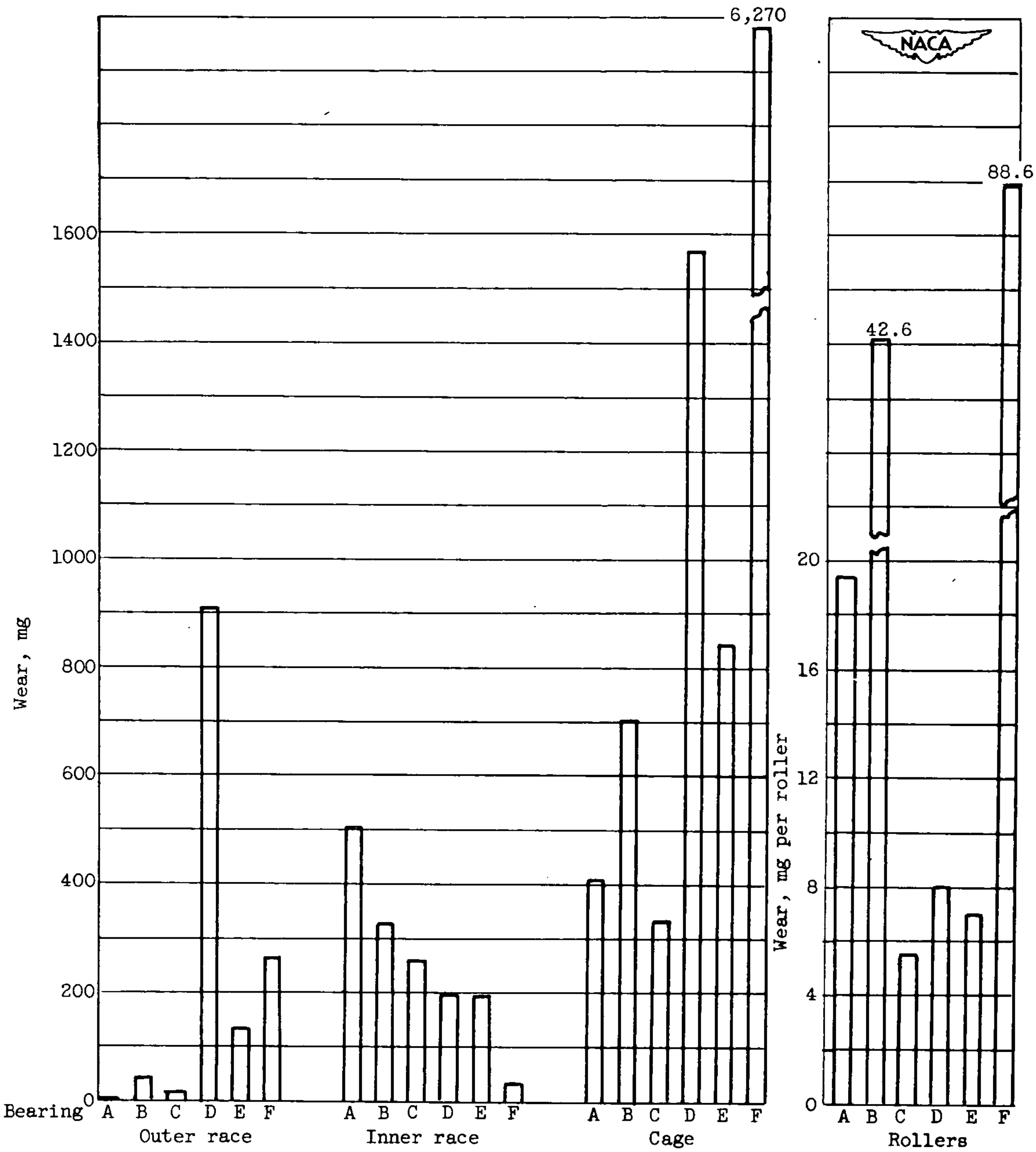


Figure 10. - Wear of component parts of bearings A, B, C, D, E, and F. (Data for bearing F from ref. 10.)

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